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Description Of The Preferred Embodiment

[0020] In the following figures the same reference numerals will be used to identify the same components. The present invention is preferably used in conjunction with a rollover control system for a vehicle. However, the present invention may also be used with a deployment device such as airbag or roll bar. The present invention will be discussed below in terms of preferred embodiments relating to an automotive vehicle moving in a three-dimensional road terrain.

[0021] Referring to Figure 1, an automotive vehicle 10 with a safety system of the present invention is illustrated with the various forces and moments thereon during a rollover condition. Vehicle 10 has front right and front left tires 12a and 12b and rear right tires 13a and left rear tires 13b respectively. The vehicle 10 may also have a number of different types of front steering systems 14a and rear steering systems 14b including having each of the front and rear wheels configured with a respective controllable actuator, the front and rear wheels having a conventional type system in which both of the front wheels are controlled together and both of the rear wheels are controlled together, a system having conventional front steering and independently controllable rear steering for each of the wheels or vice versa. Generally, the vehicle has a weight represented as Mg at the center of gravity of the vehicle, where $g = 9.8\,m/s^2$ and M is the total mass of the vehicle.

[0022] As mentioned above, the system may also be used with active/semi-active suspension systems, anti-

roll bar or other safety devices deployed or activated upon sensing predetermined dynamic conditions of the vehicle.

[0023] The sensing system 16 is coupled to a control system 18. The sensing system 16 preferably uses a standard yaw stability control sensor set (including lateral accelerometer, yaw rate sensor, steering angle sensor and wheel speed sensor) together with a roll rate sensor and a longitudinal accelerometer. The various sensors will be further described below. The wheel speed sensors 20 are mounted at each corner of the vehicle, and the rest of the sensors of sensing system 16 are preferably mounted directly on the center of gravity of the vehicle body, along the directions x,y and z shown in Figure 1. As those skilled in the art will recognize, the frame from b_1, b_2 and b_3 is called a body frame 22, whose origin is located at the center of gravity of the car body, with the b_1 corresponding to the x axis pointing forward, b_2 corresponding to the y axis pointing off the driving side (to the left), and the b_3 corresponding to the z axis pointing upward. The angular rates of the car body are denoted about their respective axes as ω_x for the roll rate, ω_y for the pitch rate and ω_z for the yaw rate. The present invention calculations preferably take place in an inertial frame 24 that may be derived from the body frame 22 as described below.

[0024] The angular rate sensors and the accelerometers are mounted on the vehicle car body along

the body frame directions b_1, b_2 and b_3, which are the $x-y-z$ axes of the vehicle's sprung mass.

[0025] The longitudinal acceleration sensor is mounted on the car body located at the center of gravity, with its sensing direction along b_1-axis, whose output is denoted as a_x. The lateral acceleration sensor is mounted on the car body located at the center of gravity, with its sensing direction along b_2-axis, whose output is denoted as a_y.

[0026] The other frame used in the following discussion includes the road frame, as depicted in Figure 1. The road frame system $r_1r_2r_3$ is fixed on the driven road surface, where the r_3 axis is along the average road normal direction computed from the normal directions of the four tire/road contact patches.

[0027] In the following discussion, the Euler angles of the body frame $b_1b_2b_3$ with respect to the road frame $r_1r_2r_3$ are denoted as $\theta_{xbr}, \theta_{ybr}$ and θ_{zbr}, which are also called the relative Euler angles.

[0028] The present invention estimates the relative Euler angles θ_{xbr} and θ_{ybr} based on the available sensor signals and the signals calculated form the measured values.

[0029] Referring now to Figure 2, roll stability control system 18 is illustrated in further detail having a controller 26 used for receiving information from a number of sensors which may include a yaw rate sensor 28, a speed sensor 20, a lateral acceleration

sensor 32, a roll rate sensor 34, a steering angle sensor 35, a longitudinal acceleration sensor 36, a pitch rate sensor 37 and steering angle position sensor 39.

[0030] In the preferred embodiment only two axial rate sensors are used. When two of these axial rates are known, the other may be derived using other commonly available sensors.

[0031] That is, pitch rate sensor 37 is illustrated, it can be eliminated in the preferred embodiment.

[0032] In the preferred embodiment the sensors are located at the center of gravity of the vehicle. Those skilled in the art will recognize that the sensor may also be located off the center of gravity and translated equivalently thereto.

[0033] Lateral acceleration, roll orientation and speed may be obtained using a global positioning system (GPS). Based upon inputs from the sensors, controller 26 may control a safety device 38. Depending on the desired sensitivity of the system and various other factors, not all the sensors 28-37 may be used in a commercial embodiment. Safety device 38 may control an airbag 40 or a steering actuator or braking actuator at one or more of the wheels 41, 42, 44, 46 of the vehicle. Also, other vehicle components such as a suspension control 48 may be used to adjust the suspension to prevent rollover.

[0034] Roll rate sensor 34 and pitch rate sensor 37 may sense the roll condition of the vehicle based on

sensing the height of one or more points on the vehicle relative to the road surface. Sensors that may be used to achieve this include a radar-based proximity sensor, a laser-based proximity sensor and a sonar-based proximity sensor.

[0035] Roll rate sensor 34 and pitch rate sensor 37 may also sense the roll condition based on sensing the linear or rotational relative displacement or displacement velocity of one or more of the suspension chassis components which may include a linear height or travel sensor, a rotary height or travel sensor, a wheel speed sensor used to look for a change in velocity, a steering wheel position sensor, a steering wheel velocity sensor and a driver heading command input from an electronic component that may include steer by wire using a hand wheel or joy stick.

[0036] The roll condition may also be sensed by sensing the force or torque associated with the loading condition of one or more suspension or chassis components including a pressure transducer in an act of air suspension, a shock absorber sensor such as a load cell, a strain gauge, the steering system absolute or relative motor load, the steering system pressure of the hydraulic lines, a tire laterally force sensor or sensors, a longitudinal tire force sensor, a vertical tire force sensor or a tire sidewall torsion sensor.

[0037] The roll condition of the vehicle may also be established by one or more of the following translational or rotational positions, velocities or accelerations of the vehicle including a roll gyro, the roll rate sensor 34, the yaw rate sensor 28, the lateral

acceleration sensor 32, a vertical acceleration sensor, a vehicle longitudinal acceleration sensor, lateral or vertical speed sensor including a wheel-based speed sensor, a radar-based speed sensor, a sonar-based speed sensor, a laser-based speed sensor or an optical-based speed sensor.

[0038] Steering control 38 may control the position of the front right wheel actuator 40, the front left wheel actuator 42, the rear left wheel actuator 44, and the right rear wheel actuator 46. Although as described above, two or more of the actuators may be simultaneously controlled. For example, in a rack-and-pinion system, the two wheels coupled thereto are simultaneously controlled. Based on the inputs from sensors 28 through 39, controller 26 determines a roll condition and controls the steering position of the wheels.

[0039] Speed sensor 30 may be one of a variety of speed sensors known to those skilled in the art. For example, a suitable speed sensor may include a sensor at every wheel that is averaged by controller 26. Preferably, the controller translates the wheel speeds into the speed of the vehicle. Yaw rate, steering angle, wheel speed and possibly a slip angle estimate at each wheel may be translated back to the speed of the vehicle at the center of gravity. Various other algorithms are known to those skilled in the art. Speed may also be obtained from a transmission sensor. For example, if speed is determined while speeding up or braking around a corner, the lowest or highest wheel speed may not be used because of its error. Also, a

transmission sensor may be used to determine vehicle speed.

Connecting The Relative Attitudes With The Relative Corner Displacements

[0040] In operation, the method according to the present invention first correlates the relative attitude with the displacement at each corner of the vehicle. Consider a vector with $x-y-z$ coordinates as x_b, y_b, z_b of its end point in the body frame of Figure 1. The z coordinator of the end point of the same vector measured in the road frame can be computed from the Euler transformation

$$z_r = -x_b \sin(\theta_{ybr}) + y_b \sin(\theta_{xbr})\cos(\theta_{ybr}) + z_b \cos(\theta_{xbr})\cos(\theta_{ybr}) \qquad (1)$$

Let l be the half of the wheel track; t_f and t_r be the distances from the center of gravity of the car body to the front and rear axles; h be the distance between the bottom of the vehicle body and the center of gravity of the vehicle along the body z-axis; θ_{xbr} and θ_{ybr} are the relative roll and pitch angles. Then in the body frame the four corners of the vehicle body where suspensions are connected with the wheel have the following coordination:

$$\begin{aligned}
&\text{LF Corner}: x = t_f,\ \ y = l,\ \ z = -h \\
&\text{RF Corner}: x = t_f,\ \ y = -l, z = -h \\
&\text{LR Corner}: x = -t_r, y = l,\ \ z = -h \\
&\text{RR Corner}: x = -t_r, y = -l, z = -h
\end{aligned} \qquad (2)$$

Let z_{lf}, z_{rf}, z_{lr} and z_{rr} be the relative displacements of the vehicle corners at the left-front, right-front, left-rear and right-rear locations, which are measured along the direction perpendicular to the average road surface. By using the transformation in Equation (1), those corner displacements relative to the road surface can be expressed as the function of the relative roll and pitch angles θ_{xbr} and θ_{ybr}

$$\begin{aligned} z_{lf} &= -t_f \sin(\theta_{ybr}) + l\sin(\theta_{xbr})\cos(\theta_{ybr}) + (z_{cg} - h)\cos(\theta_{xbr})\cos(\theta_{ybr}) \\ z_{rf} &= -t_f \sin(\theta_{ybr}) - l\sin(\theta_{xbr})\cos(\theta_{ybr}) + (z_{cg} - h)\cos(\theta_{xbr})\cos(\theta_{ybr}) \\ z_{lr} &= t_r \sin(\theta_{ybr}) + l\sin(\theta_{xbr})\cos(\theta_{ybr}) + (z_{cg} - h)\cos(\theta_{xbr})\cos(\theta_{ybr}) \\ z_{rr} &= t_r \sin(\theta_{ybr}) - l\sin(\theta_{xbr})\cos(\theta_{ybr}) + (z_{cg} - h)\cos(\theta_{xbr})\cos(\theta_{ybr}) \end{aligned} \qquad (3)$$

where z_{cg} is the relative displacement of the center of gravity of the vehicle with respect to the road surface, but measured along the body z-axis.

[0041] Referring now to Figure 3, the four equations of Equation (3) pose four constraints on a set of seven variables: z_{lf}, z_{rf}, z_{lr}, z_{rr}, z_{cg}, θ_{xbr} and θ_{ybr}. Hence a combination of three variables can be used to compute the rest of the variables. Since θ_{xbr} and θ_{ybr} are of interest in the present invention, the possible choices are choosing three variables from z_{lf}, z_{rf}, z_{lr}, z_{rr} and z_{cg} to characterize θ_{xbr} and θ_{ybr}. The direct measurement of any of z_{lf}, z_{rf}, z_{lr}, z_{rr} and z_{cg} is relatively expensive (for example, expensive laser distance sensors can be used). Hence, measuring any three of z_{lf}, z_{rf}, z_{lr}, z_{rr} and z_{cg} may be cost prohibitive in a commercial environment with current technology.

However, certain linear combinations of the four corner displacements z_{lf}, z_{rf}, z_{lr} and z_{rr} can be related to the available sensors through dynamics. When linear combinations are related to the relative roll and pitch Euler angles θ_{xbr} and θ_{ybr}, θ_{xbr} and θ_{ybr} may be characterized from the available sensor signals. In the following, the effort has been focused on finding those linear combinations of $z_{lf}, z_{rf}, z_{lr}, z_{rr}$, which bridges between θ_{xbr} and θ_{ybr}, and the available sensor signals including the lateral and longitudinal accelerations, the roll and yaw angular rates and the wheel speed sensor signals.

The Relative Attitudes Based On The Linear Combinations Of The Corner Displacements

[0042] The linear combinations of $z_{lf}, z_{rf}, z_{lr}, z_{rr}$, which serve as bridges to connect θ_{xbr} and θ_{ybr} with the available sensor signals are the following variables, which are called the relative roll and pitch gradients

$$\begin{aligned} \Theta_x &= \frac{z_{lf} - z_{rf} + z_{lr} - z_{rr}}{4l} \\ \Theta_y &= \frac{z_{lf} + z_{rf} - z_{lr} - z_{rr}}{2(t_f + t_r)} \end{aligned} \qquad (4)$$

Θ_x and Θ_y is related to the relative roll and pitch attitudes by manipulating the equations in (3). The final formula for the relative pitch Euler angle is

$$\theta_{ybr} = \sin^{-1}\{\Theta_y\} \qquad (5)$$

and the final formula for the relative roll Euler angle θ_{xbr} is

$$\theta_{xlr} = \sin^{-1}\{\frac{\Theta_x}{\cos(\theta_{ylr})}\} \qquad (6)$$

[0043] On the other hand Θ_x and Θ_y can be further related to the available sensor signals through dynamic equations which describe the vehicle body dynamics. Θ_x and Θ_y will be first broken into two portions, and related to the sensor signals.

Roll And Pitch Gradients Due To Suspension And Wheel Motions

[0044] As shown in Figure 4, a portion of the left front wheel 12b and suspension 52 are illustrated, z_{lf} can be further expressed as the sum of the two parts: the suspension stroke s_{lf} as measured by sensor 50 and the wheel displacement w_{lf} with respect to the road surface along the direction perpendicular to the road surface. The same is true for the rest of the corner locations. The sensor 50 may measures the change in the distance from the vehicle body to the wheel. If the four suspension strokes are s_{lf}, s_{rf}, s_{lr} and s_{rr}, and the four wheel vertical motions are w_{lf}, w_{rf}, w_{lr} and w_{rr}, then:

$$\begin{aligned} z_{lf} &= s_{lf} + w_{lf} \\ z_{rf} &= s_{rf} + w_{rf} \\ z_{lr} &= s_{lr} + w_{lr} \\ z_{rr} &= s_{rr} + w_{rr} \end{aligned} \qquad (7)$$

[0045] The relative roll and pitch gradients Θ_x and Θ_y may be broken into pieces according to the suspension

motion and the wheel vertical motion. The roll and the pitch gradients Θ_{x-susp} and Θ_{y-susp} due to suspension motions s_{lf}, s_{rf}, s_{lr} and s_{rr} may be defined as:

$$\begin{aligned}\Theta_{y-susp} &= \frac{s_{lf}+s_{rf}-s_{lr}-s_{rr}}{2(t_f+t_r)}\\ \Theta_{x-susp} &= \frac{s_{lf}-s_{rf}+s_{lr}-s_{rr}}{4l}\end{aligned} \qquad (8)$$

and the roll and pitch gradients Θ_{x-whl} and Θ_{y-whl} due to the wheel vertical motion defined as:

$$\begin{aligned}\Theta_{y-whl} &= \frac{w_{lf}+w_{rf}-w_{lr}-w_{rr}}{2(t_f+t_r)}\\ \Theta_{x-whl} &= \frac{w_{lf}-w_{rf}+w_{lr}-w_{rr}}{4l}\end{aligned} \qquad (9)$$

Then

$$\begin{aligned}\Theta_y &= \Theta_{y-susp}+\Theta_{y-whl}\\ \Theta_x &= \Theta_{x-susp}+\Theta_{x-whl}\end{aligned} \qquad (10)$$

[0046] The relative Euler angles θ_{xbr}, and θ_{ybr} can be also written as two parts:

$$\begin{aligned}\theta_{ybr} &= \sin^{-1}\{\Theta_{y-susp}+\Theta_{y-whl}\}\\ \theta_{xbr} &= \sin^{-1}\{\frac{\Theta_{x-susp}+\Theta_{x-whl}}{\cos(\theta_{ybr})}\}\end{aligned} \qquad (11)$$

[0047] Since there are no restrictions in Equation (11), it is valid regardless of if the four wheels of the vehicle contact the road surface or lift from the road, as soon as the accurate characterization of the roll and pitch gradients Θ_{x-susp} and Θ_{y-susp}, and Θ_{x-whl} and Θ_{y-whl} are available. Hence in the following

Θ_{x-susp}, Θ_{y-susp}, Θ_{x-whl} and Θ_{y-whl} may be computed based on the available sensor signals.

Estimate The Roll And Pitch Gradients

[0048] From the formula in Equation (8), the roll and pitch gradients Θ_{x-susp} and Θ_{y-susp} are related to the suspension stroke. The estimation schemes are sought for computing Θ_{x-susp} and Θ_{y-susp} from the available sensor signals.

[0049] Consider in Equation (3) that the distance differences between the left side corners and right side corners are equal, that is:

$$z_{lf} - z_{rf} = z_{lr} - z_{rr} \qquad (12)$$

or:

$$s_{lf} - s_{rf} = s_{lr} - s_{rr} + [w_{lr} - w_{rr} - w_{lf} + w_{rf}] \qquad (13)$$

Since the tire deflections are much smaller than the suspension stroke, from (13) it is reasonable to say

$$s_{lf} - s_{rf} \approx s_{lr} - s_{rr} \qquad (14)$$

or rewrite this as:

$$\frac{s_{lf} - s_{rf}}{s_{lr} - s_{rr}} \approx 1 \qquad (15)$$

Hence, for any given constant weight κ, we have:

$$\Theta_{x-susp} \approx \frac{\kappa(s_{lf} - s_{rf}) + s_{lr} - s_{rr}}{2l(\kappa + 1)} \qquad (16)$$

In the sequential discussion, the Equation (16) may be used to describe the roll gradient Θ_{x-susp}.

[0050] Θ_{x-susp} and Θ_{y-susp} must then be related to the available sensor signals. The following dynamic relationship which are obeyed by the car body through the Newton law described around the c.g. of the vehicle body

$$
\begin{aligned}
I_x\dot{\omega}_x &= h_y\sum_{i=1}^{4}F_{yi} + l(K_f s_{lf} + D_f\dot{s}_{lf}) - l(K_f s_{rf} + D_f\dot{s}_{rf}) + l(K_r s_{lr} + D_r\dot{s}_{lr}) - l(K_r s_{rr} + D_r\dot{s}_{rr}) \\
&\quad + K_{anti-roll-f}\frac{(s_{lf} - s_{rf})}{l} + K_{anti-roll-r}\frac{(s_{lr} - s_{rr})}{l} \\
I_y\dot{\omega}_y &= h_x\sum_{i=1}^{4}F_{xi} + t_f(K_f s_{lf} + D_f\dot{s}_{lf}) + t_f(K_f s_{rf} + D_f\dot{s}_{rf}) - t_r(K_r s_{lr} + D_r\dot{s}_{lr}) - t_r(K_r s_{rr} + D_r\dot{s}_{rr}) \\
M_s a_y &= \sum_{i=1}^{4}F_{yi} \\
M_s a_x &= \sum_{i=1}^{4}F_{xi}
\end{aligned}
\tag{17}
$$

where I_x and I_y are the momentum of inertia of the car body with respect to the x and y axis respectively; M_s is the sprung mass (the mass of the car body); h_y is the c.g. height of the car body with respect to the top of the suspension; K_f and K_r are the front and rear suspension spring rates with unit N/m. $K_{anti-roll-f}$ and $K_{anti-roll-r}$ are the stiffnesses for the front and the rear anti-roll bar, with unit Nm/rad. D_f and D_r are the front and the rear suspension damper rates; F_{xi} is the ith suspension force applied to the car body along the body fixed direction b_1, and F_{yi} is the ith suspension force applied to the car body along the body fixed direction b_2.

Define a weight:

$$\kappa = \frac{l^2 K_f + K_{anti-roll-f}}{l^2 K_r + K_{anti-roll-r}} \qquad (18)$$

Since the damping rates are usually proportional to the spring rates for suspensions, it is reasonable to assume:

$$\frac{l^2 D_f + D_{anti-roll-f}}{l^2 D_r + D_{anti-roll-r}} \approx \kappa \qquad (19)$$

For a well balanced vehicle, the normal dead loading applied to the vehicle should not generate significant body attitude variation when the vehicle is parked on a flat road. That is, the roll and pitch attitude angles induced by the normal dead loading during flat road parking should be close to zero. For this reason, it is reasonable to assume the following holds:

$$t_r K_r = t_f K_f \qquad (20)$$

Similar argument can be used for suspension damping rates.

[0051] Through algebraic manipulation the first two equations in Equation (17) can be rewritten as the following:

$$I_x\dot{\omega}_x = h_y M_s a_y + (lK_r + \frac{K_{anti-roll-r}}{l})[\kappa(s_{lf} - s_{rf}) + (s_{lr} - s_{rr})]$$
$$+ (lD_r + \frac{D_{anti-roll-r}}{l})[\kappa(\dot{s}_{lf} - \dot{s}_{rf}) + (\dot{s}_{lr} - \dot{s}_{rr})] \qquad (21)$$
$$I_y\dot{\omega}_y = h_x M_s a_x + t_r K_r(s_{lf} + s_{rf} - s_{lr} - s_{rr}) + t_r D_r(\dot{s}_{lf} + \dot{s}_{rf} - \dot{s}_{lr} - \dot{s}_{rr})$$

Using the definition of Θ_{x-susp} and Θ_{y-susp}, Equation (21) can be rewritten as:

$$\dot{\omega}_x = c_0 a_y + c_1\Theta_{x-susp} + c_2\dot{\Theta}_{x-susp}$$
$$\dot{\omega}_y = d_0 a_x + d_1\Theta_{y-susp} + d_2\dot{\Theta}_{y-susp} \qquad (22)$$

that is, $\Theta_{x-susp}(t)$ and $\Theta_{y-susp}(t)$ obeys the 1st order differential equations, and the coefficients c_0, c_1, c_2, d_0, d_1 and d_2 can be obtained by comparing Equation (21) and Equation (22). Although the analytical solution for Equation (22) are not hard to find, the solutions may be directly implemented in digital environment. On the other hand, the pitch rate signal is not measured, but an estimation of the pitch rate signal can be obtained as a function of the measured signals and the signals computed from the measured signals:

$$\hat{\omega}_y = \dot{\hat{\theta}}_y \sec(\hat{\theta}_x) + \omega_z \tan(\hat{\theta}_x) \qquad (23)$$

where $\hat{\theta}_x$ and $\hat{\theta}_y$ are the estimated global roll and pitch Euler angles of the vehicle body (with respect to the sea level). The details if this are described in U.S. Application 09/967,938 which is incorporated by reference herein. Using the estimated pitch rate signal, (22) can be used to solve for $\Theta_{x-susp}(t)$ and $\Theta_{y-susp}(t)$ at time instant t. In the following a digital scheme

will be summarized. Two variables as defined at each sampling instant:

$$RRA\_RAW(k) = \frac{1}{c_1}\dot{\omega}_x(k) - \frac{c_0}{c_1}a_y(k)$$
$$RPA\_RAW(k) = \frac{1}{d_1}\dot{\omega}_y(k) - \frac{d_0}{d_1}a_x(k) \quad (24)$$

Then at the (k+1)th sampling instant (current values), the estimates of the roll and pitch gradients $\hat{\Theta}_{x-susp}(k+1)$ and $\hat{\Theta}_{y-susp}(k+1)$ may be computed from their values in the kth sampling instant (past values) and the current and past values of RRA_RAW and RPA_RAW. The iterative formula may be expressed as the following with properly chosen coefficients e_0, e_1, f_0 and f_1:

$$\hat{\Theta}_{x-susp}(k+1) = e_0\hat{\Theta}_{x-susp}(k) + e_1[RRA\_RAW(k+1) + RRA\_RAW(k)]$$
$$\hat{\Theta}_{y-susp}(k+1) = f_0\hat{\Theta}_{y-susp}(k) + f_1[RPA\_RAW(k+1) + RPA\_RAW(k)] \quad (25)$$

[0052] The wheel motion-induced roll and pitch gradients are usually much smaller than the suspension motion induced gradients due to the small tire deflections at each wheel/tire assembly. Therefore:

$$\Theta_{x-whl} << \Theta_{x-susp}$$
$$\Theta_{y-whl} << \Theta_{y-susp} \quad (26)$$

or say:

$$\Theta_x \approx \Theta_{x-susp}$$
$$\Theta_y \approx \Theta_{y-susp} \quad (27)$$